CHAPTER 21
COMBINED CYCLES

THE NONIDEAL BRAYTON CYCLE

The Brayton cycle with fluid friction is shown in Figure 21.1 by area 1-2-3-4.

$$\eta_c = \text{compressor polytropic efficiency} = \frac{\text{ideal work}}{\text{actual work}}$$
$$= \frac{h_{2s} - h_1}{h_2 - h_1}$$

If we assume constant specific heats

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$

and

$$\eta_T = \text{turbine polytropic efficiency} = \frac{\text{actual work}}{\text{ideal work}}$$
$$= \frac{h_3 - h_4}{h_3 - h_{4s}}$$

and for constant specific heats

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

The net power of the cycle is

$$\dot{W}_n = \text{power of turbine} - |\text{power of compressor}|$$

For constant specific heats

$$\dot{W}_n = \dot{m}c_p\,[(T_3 - T_4) - (T_2 - T_1)] \tag{21.1}$$

or

$$\dot{W}_n = \dot{m}c_p\left[(T_3 - T_{4s})\,\eta_T - \frac{T_{2s} - T_1}{\eta_c}\right] \tag{21.2}$$

This equation can be written in terms of the initial temperature T_1, a chosen metallurgical limit T_3, and the compressor and turbine efficiencies [Eqs. (21.1) and (21.2)] to give

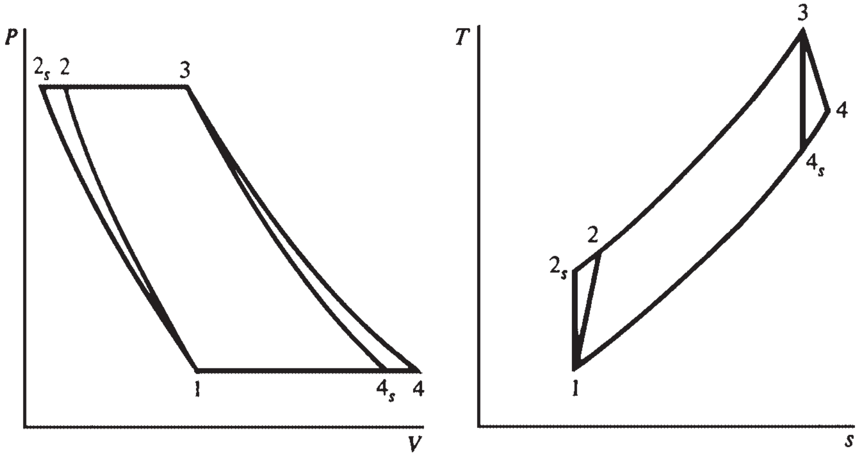


FIGURE 21.1 *P-V* and *T-s* diagrams of ideal and nonideal Brayton cycle.

$$\dot{W}_n = \dot{m}c_pT_1\left[\left(\eta_T\frac{T_3}{T_1} - \frac{r_p^{(k-1)/k}}{\eta_c}\right)\left(1 - \frac{1}{r_p^{(k-1)/k}}\right)\right] \tag{21.3}$$

The second quantity in parentheses is the efficiency of the corresponding ideal cycle.

As in the case of the ideal cycle, the specific power of the nonideal cycle, $\dot{W}_n/\dot{m}$, reaches a maximum value at some optimum pressure ratio. The heat added in the cycle is given by:

$$\dot{Q}_A = \dot{m}c_p(T_3 - T_2) = \dot{m}c_p\left[(T_3 - T_1) - \left(T_1\frac{r_p^{(k-1)/k} - 1}{\eta_c}\right)\right] \tag{21.4}$$

The efficiency of the nonideal cycle can be obtained by dividing Eq. (21.3) by Eq. (21.4). Although the efficiency of the ideal cycle is independent of cycle temperatures, the efficiency of the nonideal cycle is very much a function of the cycle temperatures. The efficiency of the nonideal cycle reaches a maximum value at an optimum pressure ratio. The two optimum pressure ratios, for specific power and for efficiency, have different values. Therefore, a compromise in design is necessary.

Other irreversibilities (e.g., fluid friction in heat exchangers, piping, etc.) have not been included in Fig. 21.1. There is a pressure drop between points 2 and 3. Also, the pressure at point 4 is greater than at point 1. Further irreversibilities occur due to bearings friction and auxiliaries, heat losses from combustion chambers, and air bypass to cool the turbine blades.

Figure 21.2 illustrates the calculation results for efficiency and specific work of a simple cycle (solid lines) and one with a regenerator (dashed lines).

The following data were used for the simple cycle:

T_1 = 15°C = 59°F = constant

P_1 = 1.013 bar = 1 atm = constant

η_c = 90%; η_T = 87%

Mechanical losses = 1%

Combustion chamber losses = 2%

Air bypass = 3%

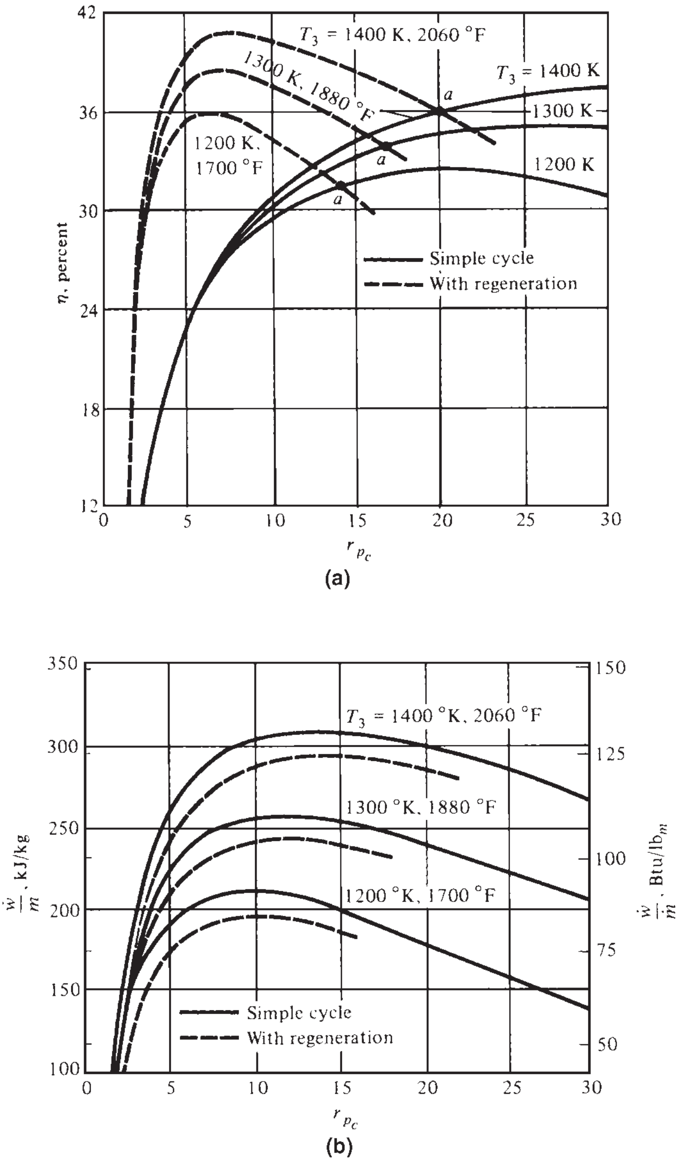


FIGURE 21.2 (*a*) Efficiency versus compressor pressure ratio of a nonideal Brayton cycle, showing effects of maximum temperature and regeneration. (*b*) Specific power versus compressor pressure ratio of a nonideal Brayton cycle, showing effects of maximum temperature and regeneration. [*Source: El-Wakil, M. M. (Ref. 1).*]

Pressure losses:

At inlet = 1%

In combustion chamber = 3%

At outlet = 2%

In regeneration = 4%

Actual, variable properties of air and combustion gases were used. Figure 21.2 indicates that the efficiency and specific work depends strongly on the maximum temperature *T,* which occurs at the inlet to the turbine.

Figure 21.3 illustrates a single-shaft, direct-cycle, open-air combustion gas turbine. A 16-stage axial compressor, 1 of 10 combustion chambers, and a 3-stage turbine are shown. A diesel engine for starting is shown on the left.

The power plant, General Electric model MS-6001, produces 35.75 MW, is 30.50 percent efficient, runs at 5100 r/min, and has overall dimensions, including electric generator (not shown), of 38 m (122 ft) long, 11 m (36 ft) high, and 8 m (26 ft) wide.

MODIFICATIONS OF THE BRAYTON CYCLE

The simple gas turbine cycle is economically adequate for peaking units and jet transport. However, base-loaded units require modifications to improve their efficiency. Some modifications required, besides increasing the combustor outlet temperature, include the following:

- Regeneration
- Compressor intercooling
- Turbine reheat
- Water injection

Regeneration

Regeneration is the internal exchange of heat within the cycle. The turbine outlet temperature is usually higher than the compressor outlet temperature. Figure 21.4 illustrates the flow and *T-s* diagrams of a closed, nonideal Brayton cycle with regeneration. The compressed gas at point 2 is preheated by the exhaust gases at point 4 in a heat exchanger called a *regenerator,* sometimes *recuperator.*

If the regenerator were 100 percent effective, the gas temperature entering the combustor would be raised from T_2 to $T_{2''}$ (T_4). The heat added would be reduced from $H_3 - H_2$ to $H_3 - H_{2''}$. In reality, the compressed gas is heated to $T_{2'}$ because the regenerator effectiveness is always less than 100 percent.

The regenerator effectiveness, ε_R, is:

$$\varepsilon_R = \frac{T_{2'} - T_4}{T_4 - T_2} \tag{21.5}$$

Figure 21.2 (*a, b*) shows the effect of adding a regenerator with $\varepsilon_R = 0.75$.

There is a significant increase in efficiency. However, the optimum pressure ratio for efficiency shifts to lower values. This is because as the pressure ratio decreases, the

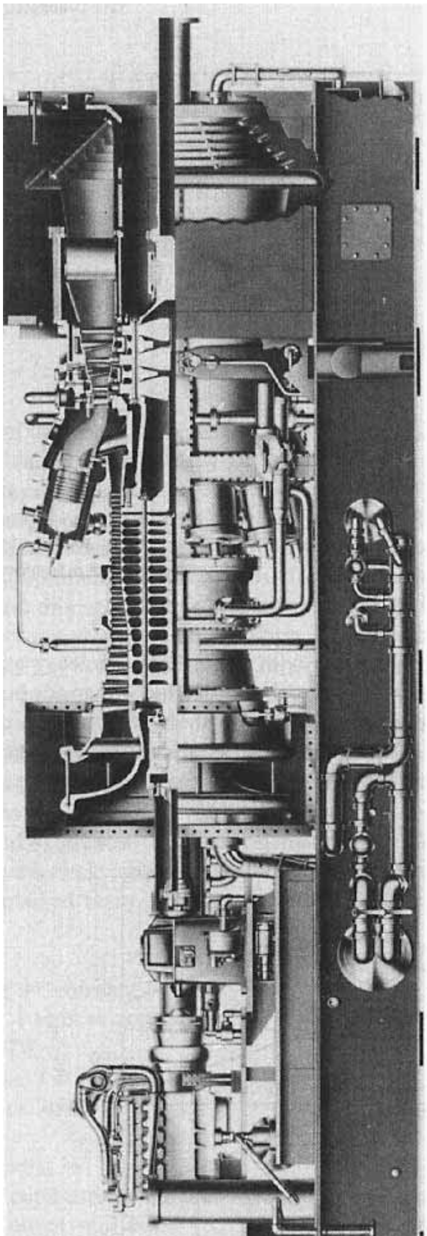
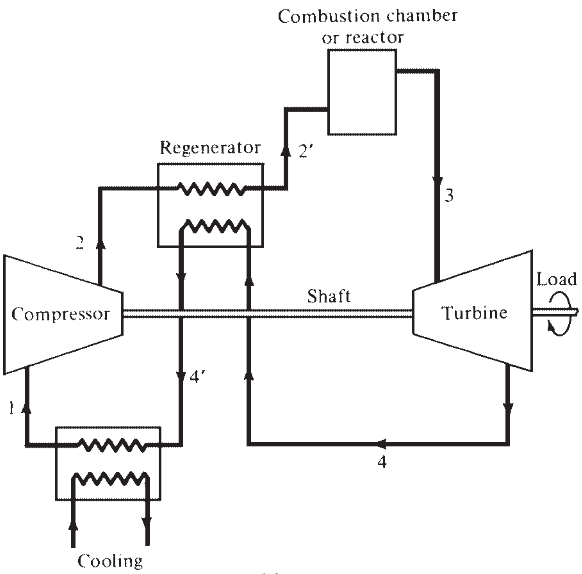
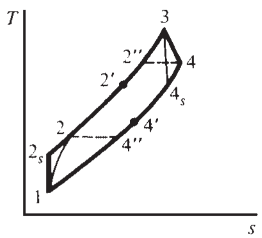


FIGURE 21.3 35.75-MW direct-cycle gas-turbine powerplant. (*Courtesy Gas Turbine Division, General Electric Company, Schenectady, New York.*)



(*a*)



(*b*)

FIGURE 21.4 (*a*) Flow and (*b*) *T-s* diagrams of a closed, nonideal Brayton cycle with regeneration.

difference between T_4 and T_2 increases. This results in a greater reduction in cycle heat input.

At a very low pressure ratio (r_p), the effect of reduced cycle work predominates and the efficiency drops significantly. The efficiency curves for a cycle with regenerator cross the simple-cycle curves at points such as point *a*. This is the point beyond which the effect of a regenerator on efficiency is negative. These points represent pressure ratios at which the turbine exhaust gases temperature (T_4) is lower than those after compression (T_2).

Compressor Intercooling

The work in a compressor or a turbine is given by:

$$W = -\int_1^2 V\,dP \tag{21.6}$$

For a perfect gas where $PV = mRT$, this equation can be written as

$$W = -\int_1^2 mRT \frac{dP}{P} \tag{21.7}$$

For a given dP/P, the work is directly proportional to temperature. A compressor working between points 1 and 2 would expend more work as the gas approaches point 2. Therefore, it is advantageous to keep T as low as possible while reaching P_2.

Figure 21.5 shows two stages of intercooling. There is a net increase in work and efficiency. The increase in work is given by area 2-1′-2′-1″-2″-x-2. The heat added has also increased by $h_x - h_{2''}$. However, there is a net improvement in efficiency.

Turbine Reheat

The turbine work can be increased by keeping the gas temperature in the turbine as close as possible to the turbine inlet temperature, T_3. Figure 21.5 shows one stage of reheat. The

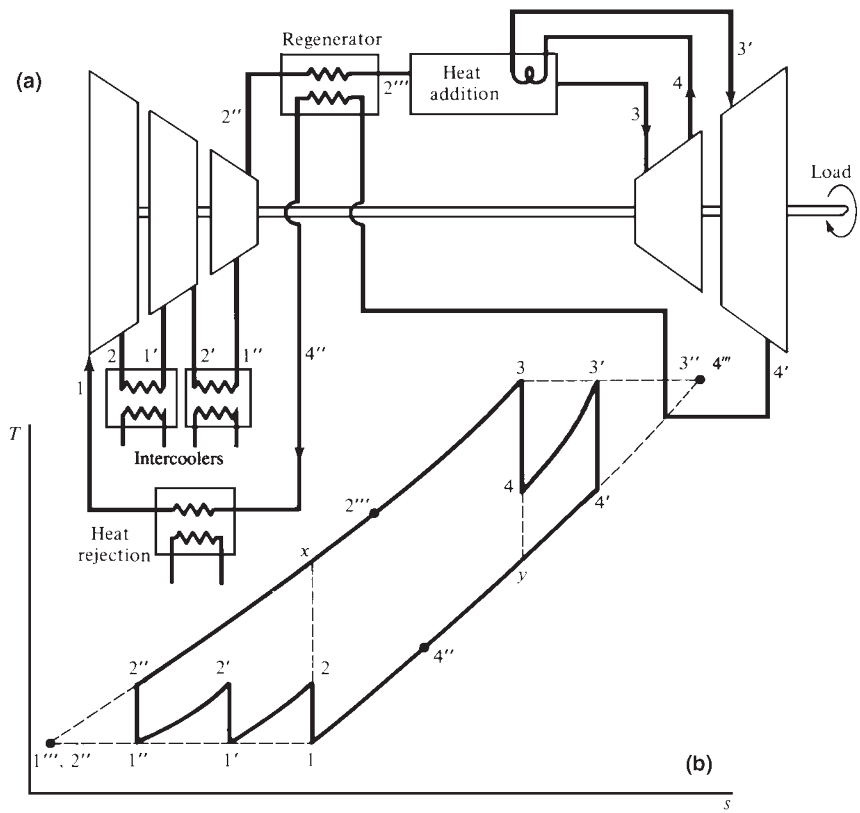


FIGURE 21.5 (a) Flow and (b) T-s diagrams of a closed, ideal Brayton cycle with two stages of intercooling, one stage of reheat, and regeneration.

increase in cycle work is given by area 4-3′-4′-*y*. The heat added has increased by $H_{3'} - H_4$. However, there is a net increase in work and efficiency. The efficiency increases when the number of reheat and intercooling stages increases. However, the capital investment and plant size would increase.

Water Injection

Water injection is a method used to increase the power output of a gas turbine significantly and to have marginal increase in efficiency. In some aircraft propulsion units, water is injected into the compressor. It evaporates when the air temperature rises through the compression process. The heat of vaporization reduces the compressed air temperature, resulting in a decrease in compressor work.

Figure 21.6 (*a, b*) shows flow and *T-s* diagrams of a gas turbine cycle with water injection and regeneration. Area 1-2-4-5-7-9′-1 represents the cycle without water injection. Point 9′ represents the exhaust gas at the outlet of the regenerator.

When water in injected, the compressed air at point 2 is cooled at nearly constant pressure by the evaporating water at point 3. The regenerator preheats the compressed air at point 3 to point 4. The added heat required to increase the temperature of the moist air from point 3 to point 2 is obtained from the exhaust gases between points 9′ and 9.

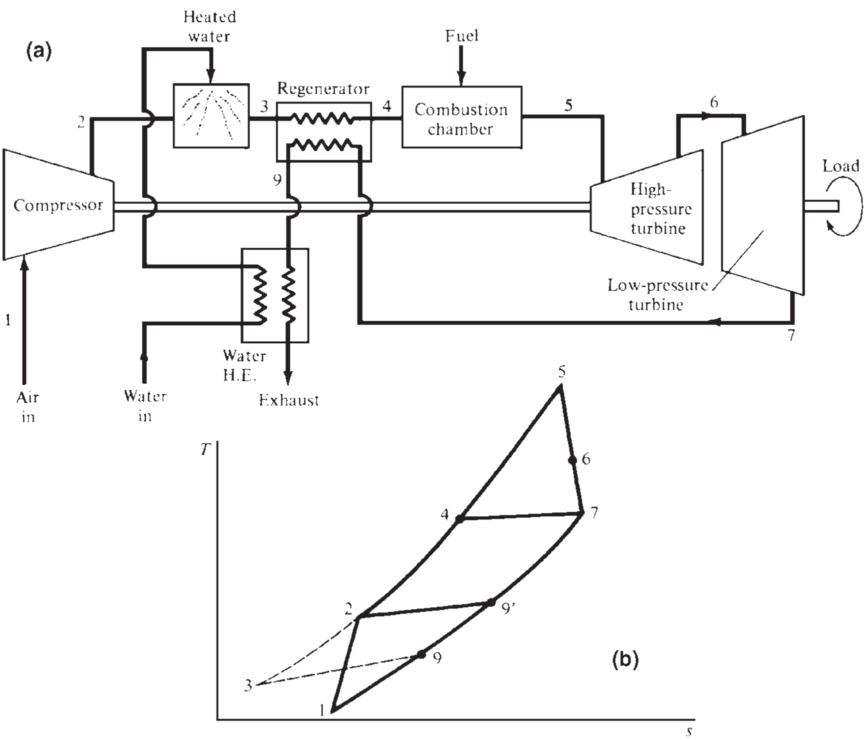


FIGURE 21.6 (*a*) Flow and (*b*) *T-s* diagrams of a two-shaft gas turbine cycle with water injection and regeneration.

The quantity of water vapor injected is enough to saturate the air at T_3. Any further increase in the quantity of water vapor would reduce the efficiency and increase the net work. The increase in water will lead to fouling in the regenerator, local severe temperature differences, and associated thermal stresses. The increase in work due to water injection is due to an increase in turbine work caused by the increased mass flow rate.

DESIGN FOR HIGH TEMPERATURE

Higher efficiencies can be achieved by increasing the turbine inlet temperature. The optimum pressures increase with increasing turbine inlet temperatures for both efficiency and power. The potential for corrosion increases with higher temperatures.

A turbine inlet temperature of between 2800 and 3000°F (1540 and 1650°C) has been reached.

These temperatures are significantly higher than the temperature at the inlet of the modern steam turbine, which are between 1000 and 1200°F (540 and 650°C).

Materials

The turbine first-stage blades (fixed and moving) suffer most from a combination of high temperatures, high stresses, and chemical attack. They must resist corrosion, oxidation, and thermal fatigue. The two recent advances are heat-resistant material and precision casting. They are largely attributable to aircraft engine developments.

The turbine first-stage fixed blades are made of cobalt-based alloys. These alloys are being supplemented by vacuum-cast nickel-based alloys, which are strengthened through solution- and precipitation-hardened heat treatment. The moving blades are made of cobalt-based alloys and high chromium content. Ceramic materials have been used for the turbine inlet fixed blades. However, problems were encountered due to inherent brittleness.

Cooling

Early gas turbines operated without any cooling. Operation at high temperatures in modern gas turbines requires cooling. The thermal stresses in the turbine moving blades are caused by the following:

- High rotational speeds
- Uneven temperature distributions in the different blade cross sections
- Static and pulsating gas forces that may result in dangerous vibrational stresses
- Load changes, start-up and shutdown

Therefore, the thermal stresses are caused by steady-state, as well as transient, operation. The transient operation will lead to low-cycle fatigue. This reduces the blade life significantly. Additional problems are encountered due to creep rupture, high-temperature corrosion, and oxidation.

In general, the blade surfaces should be kept below 1650°F (900°C) to reduce corrosion to an acceptable level. Cooling of the blades is done by making them hollow to allow the coolant to circulate through them. A hollow blade is lighter than a solid blade. It also has a more uniform temperature distribution than a solid blade. Air has been used as a coolant in

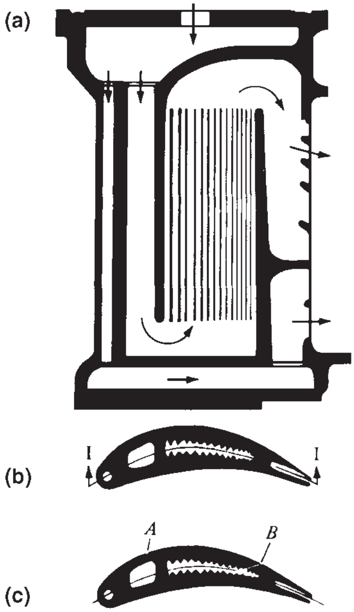


FIGURE 21.7 Air-cooled gas turbine fixed blade.

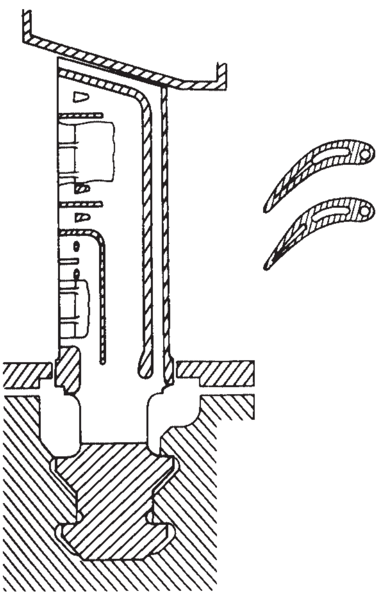


FIGURE 21.8 Air-cooled gas turbine moving blade.

gas turbines up to 2100°F (1150°C). Water has been used for gas temperature above 2400°F (1315°C).

Air Cooling. The cooling air is obtained directly from the compressor to the turbine. It bypasses the combustor.

In *convection cooling,* the air flows inside the hollow blade. It enters at the leading edge and leaves at the trailing edge to enter the main gas stream.

Film cooling is used in conjunction with convection cooling. Air flows through holes from inside the blade to the outside boundary layer to form a protective insulating film between the blade and the hot gases. This method helps prevent corrosion of the blades in addition to cooling.

Figure 21.7 illustrates air cooling of inlet fixed blades. The upper vertical cross section [Fig. 21.7 (*a*)] shows air entering from the stator at the top. It flows downward by the leading edge in two parallel paths. It changes direction a few times and leaves at the trailing edge.

Figure 21.7 (*b*) illustrates the middle horizontal cross section through the blade. It shows the internal path in pure convection cooling. Figure 21.7 (*c*) illustrates two rows of holes, *A* and *B,* on the side of the blade for film cooling.

Figure 21.8 illustrates the air cooling in the moving blades. The air enters the blade root from the rotor. It flows through the hollow blades in ducts and leaves through slots from the blade trailing edge.

Water Cooling. When the air temperature reaches 2100°F (1150°C), air cooling becomes ineffective due to the significant increase in cooling air that bypasses the combustion chamber. Water cooling is very effective when the gas temperature exceeds 2400°F (1315°C). Lower metal temperatures are reached due to the high heat transfer capability of water. It reduces hot corrosion and deposition from contaminated fuels. Water cooling also eliminates the need of passages through the blades (film cooling), which could be plugged by contamination.

Experiments using heavy ash-bearing fuels have showed lower metal tempera-

tures and reduction in ash accumulation on the blades with water cooling. The fixed blades or nozzles are hollow. They contain series of parallel flow paths. The water circulates in, through, and out of these paths in a closed loop. The heat removed from these blades is recovered in a heat exchanger for use in the steam portion of a combined cycle. The inlet water temperature is relatively high to prevent thermal shock. Its pressure is high to prevent boiling.

The moving blades are cooled by an open-loop system. The water enters the blades at lower pressures and is allowed to boil. The steam is ejected from the blade tips to mix with the gas stream.

FUELS

Liquid fuels have been used in gas turbines. However, they are viscous and form sludge when overheated. Their high carbon content leads to excessive carbon deposits in the combustion chamber. Their contents of alkali metals, such as sodium, combine with sulfur to form sulfates that are corrosive. Their metals, such as vanadium, form corrosive combustion products. They have a high ash content that deposits mainly on the inlet fixed blades, resulting in reduction in gas flow and power output.

Fuel additives, such as magnesium, have been found to neutralize vanadium. Other additives and protective coatings are also used to reduce corrosion.

The pressurized-fluidized-bed combustion (PFBC) makes coal, which is cheap, abundant, and readily available as a gas turbine fuel. In the PFBC, the addition of limestone will remove enough sulfur to meet environmental regulations. Further research is required to reduce particulate matter from the gaseous products of PFBC, which can destroy the turbine blades. Another alternative for coal usage is the use of synthetic fuels from coal gasification and liquidation.

COMBINED CYCLES

Steam and gas turbines are used to supply power in combined-cycle power plants. The idea has originated from the need to improve the Brayton cycle efficiency by utilizing the waste heat in the turbine exhaust gases.

The large quantity of energy leaving with the turbine exhaust is used to generate steam for a steam power plant. This is a suitable arrangement because the gas turbine is a relatively high-temperature machine (2000 to 3000°F, 1100 to 1650°C) while the steam turbine is a relatively low-temperature machine (1000 to 1200°F, 540 to 650°C).

Combined cycles have high efficiency, as well as high power, outputs. They are characterized by flexibility and quick part-load starting. They are also suitable for both base-load and cyclic operation, and have a high efficiency over a wide range of loads.

The most common types of combined cycles include those with heat recovery boilers (HRBs), the steam-and-water (STAG) combined-cycle power plant, and combined cycles with multipressure steam.

Combined Cycles with Heat Recovery Boiler

Figure 21.9 illustrates a schematic flow diagram of a combined cycle with an HRB. The gas turbine exhaust is going to an HRB to generate superheated steam. The HRB consists of an

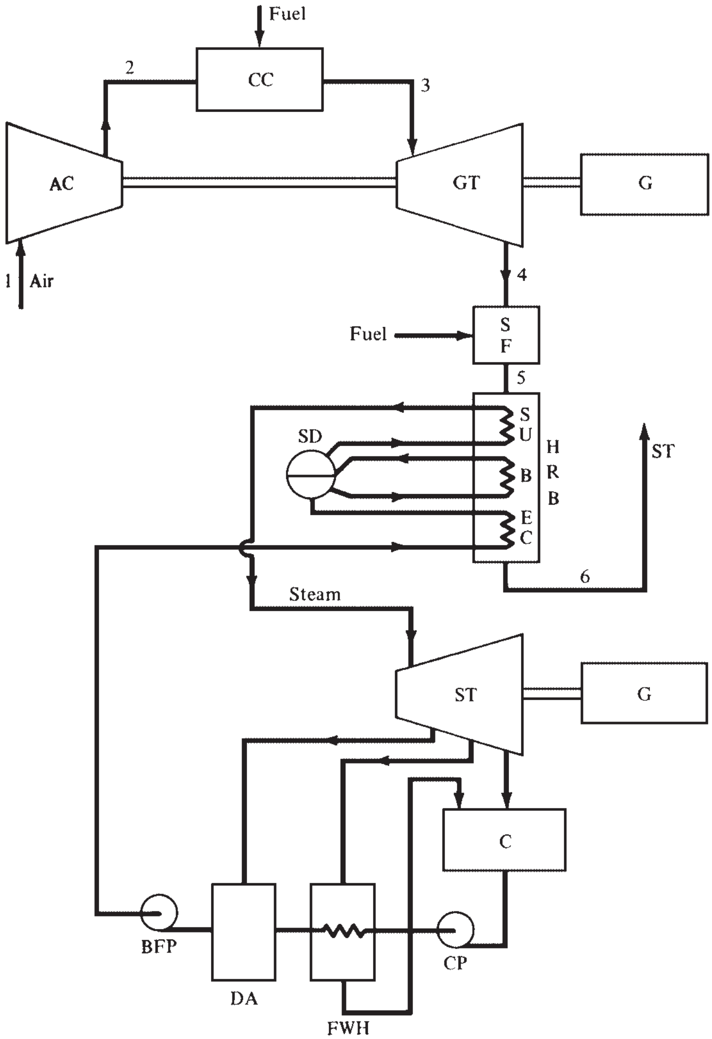


FIGURE 21.9 Schematic flow diagram of a combined cycle with a heat recovery boiler (HRB).

economizer (*EC*), boiler (*B*), steam drum (*SD*), and superheater (*SU*). The gas leaves the HRB to the stack. The gas turbine is operated with a high air-fuel ratio to make sufficient air available in the gas turbine exhaust for further combustion.

To increase the output for short periods during load peaks, supplementary fuel (SF) burners are fitted to the HRB to increase the steam mass-flow rate. This can also be done on a continuous basis.

A forced fan may be installed ahead of SF to operate the steam cycle on its own when the gas turbine is cut off. The fuel that is used in the supplementary firing can be the same

as the high-grade fuel that is used in the gas turbine or it can be lower-grade fuels, such as heavy oil or coal. However, the high-grade fuel is preferred because it causes fewer problems in the SF and HRB.

The STAG Combined-Cycle Powerplant

The steam and gas (STAG) is a 330-MW combined-cycle power plant built for the Jersey Central Power and Light Company. It is a cyclic plant designed by General Electric Company. It consists of four GE Model-7000 gas turbines exhausting to supplementary firing in the form of auxiliary burner sections within four HRBs. The HRBs provide the superheated steam to one steam turbine. Figure 21.10 illustrates the plant layout.

The STAG is an operationally flexible combined-cycle power plant. Each of the four gas turbines and the steam turbine can be started, controlled, and loaded independently from a control room. Either one or more gas turbines can be operated with its HRB supplement fired or unfired. Steam pressures of 600, 800, 1000, and 1250 pounds per square inch gauge (psig) (4.08, 5.4, 6.8, and 8.5 MPa) are obtained with one, two, three, and four gas turbines. The plant data are as follows:

Gas turbines:	Four GE Model-7000, each rated at 49.5 MW (base) and 54.9 MW (peak) at 80°F (27°C) inlet.
Turbine exhaust:	970°F (521°C). Dampers used to bypass gas to atmosphere when operating alone, or to direct gas to the HRB when operating in combined-cycle mode. Silencers are located ahead of bypass stack and HRB.
HRB:	Four single-pressure, burner-and-steam generator sections are factory-assembled modules for site erection. Forced recirculation in boiler section.
Feedwater:	267°F (130°C) at economizer inlet.
Steam:	1250 psig (87 bar), 950°F (510°C), 995,220 lb_m/h (125 kg/s).

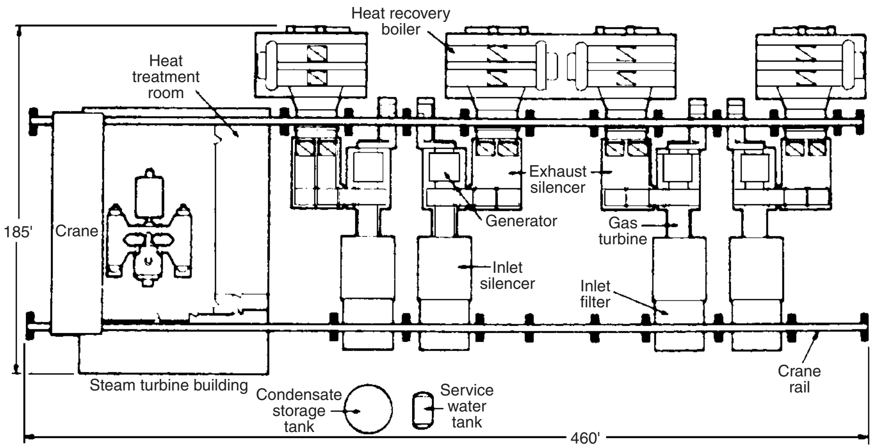


FIGURE 21.10 Layout of the STAG combined-cycle power plant.

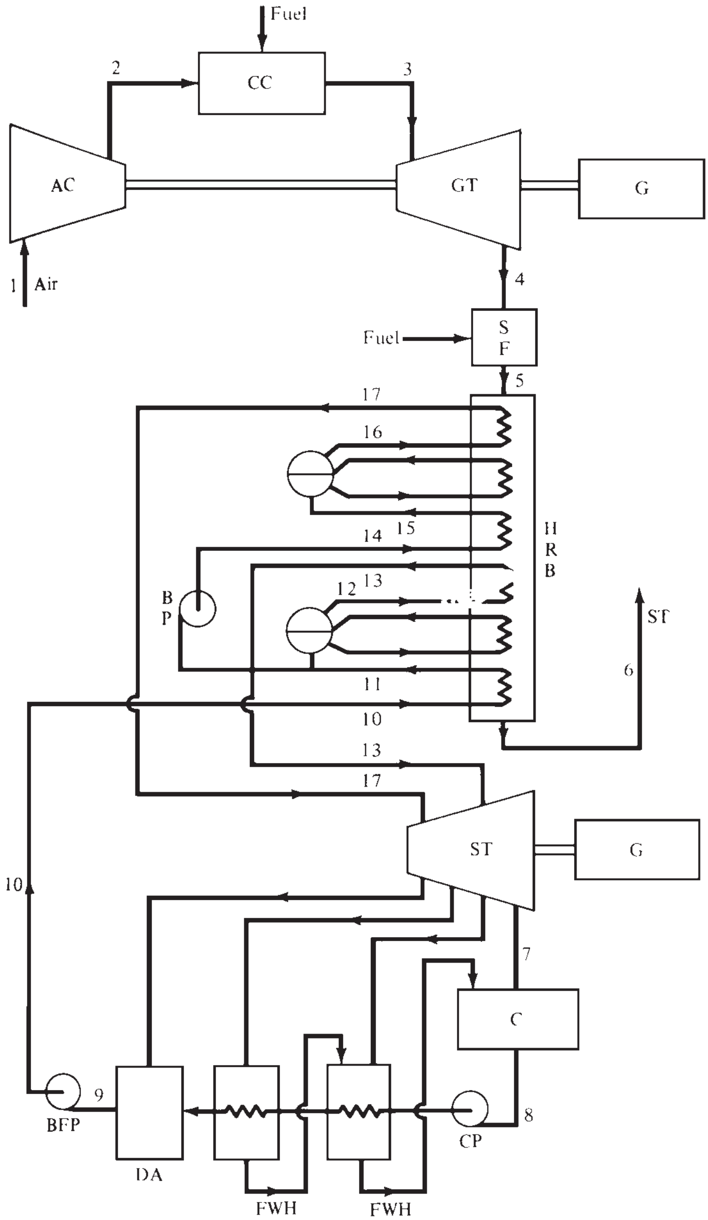


FIGURE 21.11 A schematic diagram for a dual-pressure combined cycle.

Steam turbine: One high-pressure and one double-flow, low-pressure, tandem-compound section, non-reheat, rated at 129.6 MW with 3.5 inHg (0.12 bar) back pressure.

Fuel: No. 2 distillate oil initially. Corrosion-resistant first-stage gas turbine materials allow future use of heavier fuel.

When the gas turbines are exhausting to atmosphere, the efficiency is 26.3 to 25.3 percent. When the HRB is firing and the steam turbine is at very wide-open throttle, the efficiency is 39.3 percent.

Combined Cycles with Multipressure Steam

The temperature of the gas leaving the HRB is reduced in a combined cycle having multipressure steam. This results in an increase in the efficiency of the plant. With steam cycles operating around 1300 psia (90 bar), the gas temperature leaving the HRB to the stack is around 300 to 400°F (150 to 200°C). Some of the energy leaving with the gas can be utilized in a multipressure steam cycle.

A *dual-pressure cycle* is shown in Fig. 21.11. The HRB has two steam circuits in it:

1. *High-pressure circuit.* It feeds steam to the steam turbine at its inlet.
2. *Low-pressure circuit.* It feeds steam to the turbine at a lower stage.

The corresponding temperature-enthalpy diagram of both gas and steam circuits in the HRB is shown in Fig. 21.12. Line 10-11 is a feedwater heating in a low-pressure economizer. It is followed by evaporation to point 12 and superheat to point 13. Water is pumped by a booster pump (BP) from the low-pressure steam drum at point 11 to point 14.

Figure 21.12 shows also that a single high-pressure steam circuit is represented by lines 10′-15-16-17 with the gas leaving to the stack at 6′.

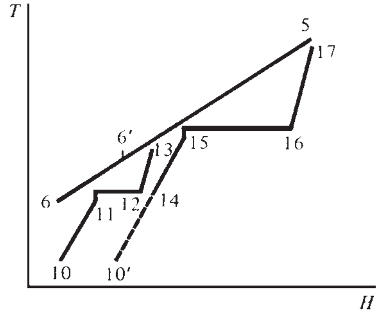


FIGURE 21.12 Temperature-enthalpy (*T-H*) diagram of the heat recovery boiler of the dual-pressure combined cycle shown in Fig. 21.11.

The addition of the low-pressure circuit allowed the gas to leave at a lower temperature (point 6). This indicates that more energy has been extracted from the gas, resulting in an increase in overall cycle efficiency. The efficiency is 46.1 percent when the air temperature is 15°C.

REFERENCES

1. El-Wakil, M. M., *Power Plant Technology,* McGraw-Hill, New York, 1984.